

in compression by tensile elements. For example, a push rod is replaced by a "pull wire." The arrangement is illustrated diagrammatically in Figure 18, with camshaft 1256 actuating rocker arm 1257 fixed at pivot 1258 which, via tensile member 1259, activates rocker 1260 anchored at pivot 1261, which in turn activates valve 1262 and spring 1263. It is clear that the use of tensile members permits greater freedom in location of cam and valve mechanism, since the line of force need no longer be a straight path. By way of example, tensile element 1259 is shown routed clear of another engine element 1264 by means of wheel, roller or bearing 1265. The rocker arrangement of Figure 18 can be eliminated, as shown in Figure 19, by attaching the tensile member 1259 to a movable cage 1266 surrounding the cam 1267, the cage having a cam follower 1268 (shown by way of example as a roller bearing) and ~~((guide))~~ slot 1269 ~~((shown schematically by way of example as a flange slidable in a slot, the latter not illustrated)))~~ moveable on a fixed cylindrical guide 1269a, for defining follower movement relative to cam in the direction indicated by arrow 1270.

A selected embodiment of the engine is illustrated schematically in Figure 20. It consists of a piston 1001 reciprocating between two combustion chambers 1002 at each end of a cylinder 1003 closed by two heads 1004, with a crankshaft 1006 outboard each head, the piston being connected by tensile members 1007 to both crankshafts. Optionally, the crankshaft will also function as a camshaft, actuating valves and optionally providing fuel delivery. The liquid elements for the charge may be delivered to the combustion chambers under pressures and temperatures higher than normal in conventional engines. The cylinder is at least partially surrounded by an exhaust gas processing volume 1008, with exhaust gas being conducted to the volume by alternate paths 1005 and 1009. Intake to the combustion chamber is via the crankcase. Surrounding the engine is a heavily thermally insulated casing 1010, here functioning as structure enclosing volume 1008. This configuration is suitable for four and two stroke embodiments, consuming fuel ranging from gasoline and similar lightweight fuels through diesel and heavier oil fuels to coal and other slurries or powders. Any engine lubrication and / or bearing system may be employed, but optionally either gas or roller needle bearings are used, perhaps with water or other liquids, in the case of water preferably when the components are of ceramic material, as described later. The crank assembly is preferably so designed that any air bearings at least partially operate at a pressure equivalent to the charge pressure of forced induction, in the case of turbocharged, supercharged or force-aspirated engines. In the case of two stroke engines, the preferred arrangement is to exhaust gases via ports about the center of the cylinder. In the two cycle form illustrated schematically in Figure 21, pressurized air is ducted via crankcase 1275 and valve 1276, actuated optionally by combined crankshaft / camshaft 1277, to combustion chamber 1288 (fuel injection system not shown), displacing exhaust gas which exits the chamber via ports 1289 to circumferential exhaust gas processing volume 1290. Insulation 1010 extends around the engine of Figure 20, and is shown around the crankcases and engine of Figure 21. In another example of either a two- or four-stroke engine, Figure 22, the ~~((cylinder))~~ schematically shown piston / cylinder module 1271 is linked to a single crankshaft 1272 by tensile elements 1273 routed about guides / bearings / rollers and / or wheels 1274.

The layout described above may be arranged in multiple cylinder form in a flat configuration, as is shown in plan Figure 23, longitudinal section Figure 24 and cross section Figure 25, where five piston / cylinder modules 1271 with ~~((cylinders and))~~ ten combustion chambers are

arranged about two crankshafts 1006 in two crankcases 1277, connected at one end to the transmission 1011 and optionally mechanically linked by it, and at the other end driving ancillary systems ((+275)) 1269 (such as a turbocharger) ((and optionally linked by member)) with the crankshafts optionally mechanically linked by system 1012. The space surrounding the cylinders can be used as an exhaust processing volume 1290, as shown in Figure 21. Figures 23 through 25 have been dimensioned in terms of unit d , in this case and being both the bore and the stroke of the piston. As previously, in following Figures 26 through 32, there are indicated schematically twin combustion chamber piston / cylinder modules at 1271, optionally thermally insulated engine casings at 1010, crankshafts or their axes at 1006, mechanical linkages for multiple crankshafts at 1012, spaces for ancillary systems or transmissions at 1275. In an alternative configuration, shown in schematic longitudinal section Figure 26 and cross-section 27, a double row ten cylinder engine is shown. Obviously, any number of rows and cylinders can be combined between two crankshafts, since it is only necessary to lengthen the tensile elements. In Figure 28 and 29, a schematic cross-section of a four row engine of eighteen cylinders and thirty-six combustion chambers is shown, where tensile members 1013 and 1014 are of unequal length. (In the embodiment of Figure 29, the outer pistons would normally have a shorter stroke than the inner pistons.) Either separate camshafts or more elaborate valve / fuel activation linkages are required, to provide valve actuation or fuel delivery for engines having three or more rows of cylinders and two crankshafts. Alternatively, more than two crankshafts can be employed, as shown diagrammatically in cross-section Figure 31 and longitudinal section Figure 30, in the case of a six row forty-two cylinder, eighty-four combustion chamber engine. It will be noted that these configurations are most practical if the engines are un-cooled or adiabatic. If the tensile members are replaced by connecting rods, a single crankshaft may be used, as shown diagrammatically in Figure 32 for a two row engine, having twin-combustion-chamber cylinder modules 1271, a single combined crank / camshaft 1015 and two camshafts 1016, various valve actuation rods 1276. Figures 23 through 32 are all schematic and do not show valve guides and springs, fuel delivery and exhaust systems, etc.

The basic cylinder modules may be combined to form a "ring" engine with the interior space optionally used for a turbine or ram jet engine to form a compound engine having a single revolving system. Schematic sections Figures 33 and 34 show three rings between outer casing 401 and inner casing 402, each of four piston / cylinder modules ((+271)) 403 linked by common crankshafts ((+272)) 404 and tensile members 405, with hot exhaust gases ((+280)) 406 providing at least partial energy for the ramjet or turbine ((+279)) 407, either directly or via heat exchangers (not shown). Ambient air flow is shown at 410. The work from the reciprocating portion of the engine - shown at zone 408 - may be used conventionally, may power the compressor of the turbine portion or may, as shown schematically at ((+281)) 409, drive a fan, propeller or Archimedes screw to provide thrust, either through air or water.

Amend the five paragraphs starting "The issue of the tensile link . . ." on page 16 as follows:

The issue of the tensile link between piston and crank is more complex than is immediately apparent. In the twin crankshaft layout described previously, it is not possible to maintain a constant length between piston and crank, if the cranks are to rotate synchronously.

Diagrammatical Figure 35 shows ~~((equal-synchronized crankshaft))~~ centers 1100 of equal and synchronized crankshafts 1098 with throw of radius r describing path 1099 rotating in the same direction 1101, shows piston 1102 and head / cylinder module 1103 of constant dimension k , solid line 1104 representing tensile member when the piston is in the middle of the cylinder, and dotted line 1105 the tensile members when the piston is at the end of the cylinder. In the latter position it will be seen that, if crank centers are placed $3r$ length on piston axis outboard of module, the total tensile length between crankshafts is $2r + 4r + k = 6r + k$. When the piston is in the center, the tensile member dimension is hypotenuse of right angle triangle base $a-c$ plus hypotenuse of right angle triangle base $d-f$ plus k . Since the bases total $6r$ and since the hypotenuses must be longer than the bases, it follows that the distance between the cranks is longest when the piston is in the middle of the cylinder. Since the components need always to be linked, the length of the tensile member is that required to accommodate the piston in or around the middle of the cylinder, meaning that there will be slack in the tensile system when the piston is towards the ends of the cylinder (or the tensile system has to be elastomeric). This slack is an important feature of the design of tensile crank link engines and is described in more detail later.

So far symmetrical situations have been considered - the same parameters apply to both of the combustion chambers of the piston. If the rotation of the cranks is not synchronous, then asymmetrical conditions maybe obtained, as shown schematically in Figure 36, where tensile members are shown in alternative configurations 1106, 1107. The piston 1102 is shown dotted at 1097 when it is in the center of the cylinder. Two optional tensile configurations "a" "a" and "b" "b" are shown when the piston is in the center of the cylinder traveling in direction 1108 compressing the fluids in chamber 1109. When the optionally linked cranks have traveled through 180° , tensile parameters have changed with respect to the identical piston now in compression relationship for combustion chamber 1110. (Obviously the cranks will complete revolution if the tensile members have the required amount of slack.) In order to better understand the principles of tensile crank design only symmetrical layouts will be considered from here on.

The tensile link may be wholly of some flexible material 1106, or may partly comprise a rod 1096, as shown schematically ~~((at a and b in Figure 37))~~ in Figures 37A and 37B. In both examples an equal portion of the tensile element is parallel to piston 1102 movement; in one case it is fixed relative to crank centers 1100, in the other it reciprocates and is relative to the piston. The tensile links are shown at 1006 in a first position relative to the piston, and at 1007 when the piston is shown in dashed position 1094. Here the cranks are shown turning in the same direction and the free portions of the tensile element are angled at 180° or less to one another. Not shown, but equally possible, is to have the cranks turn in opposite directions to one another, thereby maintaining the free tensile portions at a more or less constant 180° to one another. In Figure 38 an arrangement for differential pivots 1093 for each half of the cycle is shown, which will cause the piston 1102 to be off cylinder assembly 1103 center at 1092 in position 1091 when the ~~((cranks are))~~ crank(s) 1098 is / are 90° off BDC/ TDC, so permitting differential piston speeds during cycle phases. For example, such an arrangement could be used to cause the piston to move faster during the main portion of the compression stroke compared with during the main portion of the expansion stroke or vice-versa.

~~((Figure 39 shows at (a) and (b)))~~ Figures 39A and 39B show how the basic configuration of

a piston 1102 having a combustion chamber at each end and reciprocating in a cylinder assembly 1103 can be used for four stroke and two stroke engines respectively. ((with intake)) The intake phase is shown at 1111, compression at 1112, expansion at 1113, exhaust at 1114. Direction of piston travel is indicated by the arrow below each numbered portion of the Figure. In the case of the two cycle engine, only net loads are transferred to crank; in the case of the four cycle alternately net and gross loads are taken up, suggesting that for a given number of cylinders the two stroke will be smoother running. The base configuration of Figure 20 improves two stroke smoothness over conventional systems more than that of four cycle engines.

Referring back to Figure 35, it is assumed that when the cranks turn through the 90° relative to BDC/TDC, the piston is in the center of the cylinder and the tensile halves have equal slack. Considering one combustion chamber, by enlarging crank movement radius, the slack toward TDC will be decreased and the slack at BDC increased by a slightly greater amount. Reducing the crank radius to less than that of piston movement reverses the process - there is more slack at TDC and less at BDC. It is also obvious, Figure 35, that the greater the distance from head to crank center in proportion to crank radius, the less slack is required in the system. In some embodiments, it may be preferable to have little or no slack at TDC, since the piston as it approaches TDC may need to be pulled there by the crank to complete the compression and subsequently, as expansion takes place, the loads must be transferred as quickly as possible to the same crank. On the other hand, toward BDC all the useful work of expansion will have been completed, so a taut tensile member may not be required. In practice, to enable the tensile member to be taut at TDC, the crank movement diameter will have to be around 5/4 to 8/7 of piston movement, depending on design details. The presence of slack towards the ends of piston travel could cause it to spend more time there, allowing more time for combustion to develop and / or for fluid transfer to take place. The ratio can be reduced for equivalent crank centers, by employing the configuration of off an offset crankshaft 1098 shown schematically and exaggeratedly ((~~of~~)) in Figure 40, ((~~say in~~)) which may be suitable for low power applications where axial loads at the head do not present a particular problem. The piston rod 1096 is shown in part of the cylinder assembly 1103 when combustion chamber 1091 is at maximum expansion and the piston (not shown) is at BDC, with tensile link 1106 connected to the crankpin at "a". The link is shown dashed in alternative positions, with the crankpin at "b" when the piston is approximately at the center of its travel range, and at "c" when it is at TDC. (In those applications where the cranks may not rotate synchronously, differential rotation could be absorbed by using final drive devices such as illustrated in Figures 96 and 97.)

Amend the paragraph starting "The crankshaft itself . . ." on page 19 as follows:

The crankshaft itself may be manufactured along conventional lines and may be of any material, including ceramic. Non-conventional configurations may also be used, including the built-up configurations shown schematically in Figure 41, wherein center bearing tubes 1115 and big end bearing tubes 1116 are mounted in compression by axial tensile fasteners 1117 between discs 1118 which act as crank throws. These discs may be so formed as to both permit maximum bearing size and to allow the circumferential area to act as a cam, as shown in cross-section by way of example in Figure 42, where two shaped discs 1119 having precisely machined surface

cam profiles 1120 for valve cam follower 1121 actuation and fuel delivery cam follower 1122 actuation. The discs are interconnected by tensile fastener 1123 and inner crank bearing cylinder shell 1124 having precisely machined ends, each disc being similarly fastened to an inner main bearing cylinder shell 1125. Outer main bearing cylinder shells 1126 are attached to engine structure. Outer crank bearing (big end bearing) cylinder shells 1127 are attached to crank connecting rod or tensile member 1135. The present layout is shown having gas bearings where the largest bearing areas are desirable, but of course roller or needle bearings may also be employed. The technology of both gas and needle roller bearings in ceramic and other bearings is well understood and not itself a novel feature. If the gas bearings required greater than ambient gas pressure, then gas passages 1128 communicating with a central gas reservoir may duct gases to apertures 1129 at the bearing surfaces. In an alternative arrangement suited to ceramic materials and high crankcase temperatures (say around 450° K and over), the passages may contain water under pressure, which on leaving the apertures will instantly turn to steam, so providing gas under pressure in the relatively close tolerance (sometimes 1-3 microns) of the gas bearing. Optionally, the centers 1124a of the inner bearing shell cylinders 1124 may be filled with water to provide, together with likely counterbalances, some kind of flywheel effect. In crankshafts having few throws, the gas or liquid may be pulsed, to provide maximum pressures at moments of greatest loading. Instead of the apertures, a combination of apertures and wicks may be provided, as shown diagrammatically in longitudinal and cross-section in Figures 43 and 44, where 1123 and 1124 are respectively the inner and outer bearing shells, and 1136 the space between them for bearing fluid. A wick 1130 is disposed at maximum loading area 1131, to more evenly distribute the liquid delivered under pressure via passages 1132 and apertures 1133. In arrangements described elsewhere, the slack in the tensile element may optionally be taken up by a fluid spring, so that tautening of the tensile element causes fluid to be delivered to the bearings. The crank of Figure 42 is shown having lateral or axial motion, permitting the cam followers to be actuated to varying degree by the progressively shaped cam profile, as the crankshaft is moved in direction 1134. Here it is assumed that the link between piston and crank 1135 is not laterally movable, entailing larger inner bearing cylinders or shells than outer ones. Water lubrication is cited as an example; in fact any suitable liquid under pressure may be used, whether or not it changes to a gaseous phase in the bearing gap.

In the paragraph beginning "Figures 72 and 73 show . . ." on page 23, change Figure numbers as follows:

Figures 72 and 73 show in diagrammatic cross-section two versions of a "stretched circle" bearing which permits take up of slack, where a tensile / compressive link 1282 is integrally attached to non-circular outer bearing shell 1283. Between outer shell and inner bearing 1284 shell is a compressible substance 1285, with Figure 72 showing an intermediate shell 1286 to contain the compressible substance. The intermediate shell may be free to revolve or may be located relative to outer shell by guides, shown schematically at 1287. Any kind of compressible material may be enclosed at 1285, including elastic ceramic fiber assemblies, polymers, springs, etc. In selected embodiments, fluids are used, preferably gases. When a load is applied in direction 1289 the gap between shells at "a" will tend to reduce. If an aperture is provided at 1290 and clearance space at 1291 is minimized, then fluid under pressure will be forced through the

gap into main bearing clearance space 1292, providing bearing support. In the case of gas bearings, pressure can be made proportional to load by such means. If in Figure ((72)) 73 the compressible material is a gas and the clearance spaces are kept to a minimum at 1293, then gas pressure on working bearing faces is more or less continuously proportional to load. If it is desired to shift bearing shells rapidly in relationship to each other (the range of possible movement is shown dotted at 1294), then it is possible to provide a phased pressure relief to provide rapid shell movement. In Figure ((72)) 73 for example, the crank web disc 1295 is provided with apertures 1296 linked by passage 1297 so that as the disc turns in direction 1298, the relative angle to link 1282 changes to permit both the apertures to simultaneously communicate with volume 1288, permitting rapid gas transfer from one side of the volume to the other. As the crank continues to turn, the relative angle of 1282 changes to mask one of the apertures, and so shut off transfer of gas via the passageway. Figure 74 shows the layout of the variable radii of the interior surface of an outer gas bearing shell, provided with pressurized gas via apertures 1299, so as to permit progressively larger clearance gaps at the perimeter of contact area, as the inner bearing shell 1300 approaches the midpoint of its relative movement range. It is clear that differing interior profiles of the mid section of shell 1283 will cause varying travel speeds of inner shell 1284 between end positions, and so rates of acceleration and deceleration will be governed by varying shell profiles. The pressure in the gas bearings may be made directly proportional to the pressure in the combustion chamber (and therefore also partly proportional to the loads on the link) by means of small passages 1301 communicating with the chamber, providing gas access to the highly loaded bearing areas via apertures 1302, either on both sides of the volume (Figure 72) or on one side only (Figure 73). The passage from the combustion chamber may be interrupted by a filter or one-way valve mechanism shown schematically at 1303. A one-way pressure relief valve would permit only high pressure gases to pass in direction 1304, permitting gas bearing pressure to be higher than the combustion chamber pressure during portion of the cycle.

Amend the paragraph beginning "In, for example, the case of . . ." on page 24 as follows:

In, for example, the case of compound engines, it may be desirable to use exhaust gas at high temperature and pressure to power a turbine, and to have a requirement for exhaust pressures to be low to facilitate two stroke combustion chamber scavenging. In such cases more than one exhaust processing volume may be incorporated in an engine. Figure 75 shows a schematic cross-section of a five cylinder engine with a high pressure, high temperature exhaust volume at 1308 with exit at 1309, surrounded by a low pressure, low temperature volume at 1310 with twin exits at 1311. Figure 76 shows a schematic layout of a compound system with a reciprocating engine 1312 having ambient air intake 1313, high pressure exhaust 1314 and low pressure exhaust 1315. High pressure exhaust is conducted to a high performance turbine 1316 to exit at 1317, at a pressure approximately matching that of low pressure exhaust 1315 with which it is mixed, and be conducted through low temperature turbine 1318 to emerge at 1319 as close to ambient pressure as possible. Optionally the turbines might be linked by shaft 1320. Figure 77 shows a cross-section of the engine of Figure 75, where high pressure exhaust ports 1321, closable by non-return valves 1322, communicate with high temperature and pressure exhaust reservoir 1323. The piston 1323A when at BDC/TDC un.masks ports 1324, communicating with low temperature and pressure exhaust reservoir 1325. Thermally insulating structure 1328 encloses both

volumes 1323 and 1328. Figures 78 to 80 show a cylinder module made up of three elements, plus piston/rod assembly, valves, etc, and incorporating two exhaust processing volumes. The high pressure volume has four shaped snap-in non-return spring loaded valves 1326. Figure 78 is a long section and Figure 79 a cross section through the cylinder, while Figure 80 shows one valve 1326. The modules are assembled via tensile fasteners 1327, which also attach an evacuated thermally insulating cover 1328, separated from structural elements by trapped air space 1329. An intermediate thermally insulating partition is shown at 1328a. Modules are attached to each other via tensile fasteners 1327, with crank cover 1331 attached last at 1332. A similar construction, including tensile fasteners 1327, is shown also in Figures 68 and 69. On the expansion stroke, the gases are at sufficiently high pressure to open the non-return valves 1326. As the piston exposes the low pressure system via the central port 1324, the pressure in the chamber drops sufficiently to cause the spring loaded valves 1326 to close. On the compression stroke pressures will be much lower and insufficient to re-open the valves.

Amend the paragraph beginning "As has been disclosed elsewhere . . ." on page 27 as follows:

As has been disclosed elsewhere, cam and / or crank shafts may be supported in variable pressure gas bearings, with gas in the bearing either provided as a gas, or as a liquid conducted under pressure to the clearance space, which then changes state in the lower pressure / higher temperature environment of the clearance space. These fluid pressures may be varied during rotation by what can best be described as moving profile cams, which provide pumping action within the revolving body. In schematic cam / crank section Figure 84, two different arrangements are shown in a crank disc web 5100 with a big end bearing at 5021, having interior passages 5101 supplying bearing fluid being interrupted by reservoirs 5102, closed by movable plungers 5103. The plungers are linked to the free ends of movable pedals 5104, pivoted at disc surface 5105 and at disc perimeter plane 5106. Fixed cam followers 5107 are positioned, so that when the shaft turns in direction 5108, the pedals and therefore plungers are depressed when passing under the followers, causing a pressure wave in the bearing fluid. Such pedal and plunger arrangement can also be adapted to provide engine fuel delivery, where a revolving cam actuates a fixed pedal (not illustrated). If it is desirable that fluid pressure should vary not only with crank rotational angle but also with rotational speed, arrangements broadly similar to that shown schematically in sectional plan Figure 85 and the cross-section at "A" shown in Figure 86 can be employed. Here a pedal 5109 pivoted at 5111 is mounted on the ~~((disc))~~ circumferential face of a crank web disc 5110, ((the pivot being)) which also has a reservoir 5103 housing a movable plunger 5103 and connected to fluid supply 5114 and delivery 5115 passages. On the external surface of ~~((e))~~ the pedal a weighted shoe 5116 is slidably mounted. During rotational movement 5117 the shoe will pass under fixed cam follower 5118, causing the pedal to be depressed and ~~((created))~~ creating a pressure wave in the bearing fluid. The radial motion 5119 of the shaped shoe on the surface of the pedal is restrained by spring 5120. A stop restraining the movement of the plunger is shown schematically at 5021. As rotational speed increases centrifugal force on the shoe will cause the spring to be extended and shoe to move radially outward on the inclined pedal plane, causing the head of the shoe to project further from the disc surface, and increasing plunger motion during each pass under the follower. By such radial movement varied proportionally to centrifugal force, fluid pressure may be varied proportionately to crank revolution

speed.

Amend the paragraph beginning "The valveless embodiments easily . . ." on page 29 as follows:

The valveless embodiments easily permit the introduction of another feature (embodiable with greater complexity in valved engines): multiple varied diameter toroidal combustion chambers which are simultaneously in compression and subsequently expansion, and which are shown schematically in Figure 94. Each of the toroidal combustion chambers 2021, 2022, 2023 has the same cross section, but have different diameters. Dimension "b" represents stroke plus clearance space, while dimension "a" represents toroid external diameter minus internal diameter. Instead of the three chambers shown, it would be possible to have a single toroid (cross-section $b \times 3a$) of equivalent capacity. However, its clearance space cross-section would be $(b - cr) \times 3a$, while the cross-section of the clearance space of each of the toroids shown would be $(b - cr) \times a$; each would have clearance cross-sectional aspect ratio three times less steep than the single toroid. The stepped configurations of the two components also make it easier to design bearing surfaces of the required rigidity. The arrangement shown in Figure 94 permits the two ports 2009 and 2009a to be matched up to each other about midpoint of piston travel, for a relatively brief period relative to porting at bottom dead center (since the piston is traveling at maximum speed). This might be for the purpose of providing extra air to the exhaust, or to cool it. Figure 95 shows an arrangement where there is no such overlap or port to port alignment. Both Figures 94 and 95 are schematic and show only those combustion chambers on one "side" of the piston, that is those chambers that are synchronously all at top or bottom dead center. It is obvious from previous disclosures that additional combustion chamber(s) may be incorporated on the other "side" of the piston.

Amend the paragraph beginning "Figure 98 illustrates . . ." on page 31 as follows:

Figure 98 illustrates the fundamental principles of one such cam system. A circumferential sine shaped trench 2049 surrounds the midpoint of a piston/rod assembly 2050. In the trench is a guide 2051 fixed to the housing 2052, in such a way that all reciprocating motion is partially converted to rotational motion. Dimension a indicates the broad location of the circumferential band in which the cam system operates. Essentially the cam and trench system is a face system, in which the faces are aligned toward directions 2053. When the cam system is referred to as sine shaped it is for convenience; in fact the shape may be of any zig-zag type configuration. There are certain optimum profiles for each application, shown here within square 2054 which schematically describes one reciprocating cycle. Figure 99 shows the profile of an engine of the type disclosed in Figures 94 and 95, which has three cam systems, operating within bands a, b, c. The cam profile for one reciprocal cycle is identical for each band, but a different number of profiles or cycles are deployed in series within each circumferential band. The systems described each have a female and a male element (corresponding to trench 2049 and guide 2051 in Figure 98). In the three cam systems of Figure 99, the male elements are wholly or partly retractable, and only those of one band are engaged at any one time. Because loads are alternately transferred from one face to the other, the trench profile need not correspond exactly to the travel path of

the piston relative to the housing. As one cam system is disengaged and another engaged, the ratio of rotation relative to reciprocation changes, effectively making the device schematically shown in Figure 99 a three-speed variable transmission. The trench might have a clear path 2055 (shown in Figure 98), where a small guide will permit piston rotation without reciprocation, and / or a path 2056 which will permit piston reciprocation without rotation. Figure 100 shows schematically a guide of varying size, which may be wholly or partly retractable. It consists of a series of sliding tubes 2057 biased to a retracted position in a housing 2058 and where some hydraulic or other action projects each tube sequentially, those of smallest diameter before those of larger (with retraction effected in reverse sequence). If the smallest form of such a guide is able to describe a clear path in the trench, the arrangement of Figure 99 can be accomplished by having the smallest form of each of the guides of all three cam systems extended, with selective and / or progressive enlargement of the guides of only one cam system to effect rotation. It is intended that cam systems that have an adjustable portion (eg a retractable guide) may also be used to function as clutches. Without engagement, the engine would only reciprocate; with cam system engagement rotation commences. In the case of guides which are rollers, it may be preferable to have them tapered, with correspondingly inclined cam faces. Figure 101 shows a schematic cross-section through a piston 2059 having axis of rotation at 2060. Two rollers 2061 are fixedly mounted to housing 2062 and rotate about axes 2063 when engaged in channel 2064.

Amend the paragraph beginning "For certain applications . . ." on page 33 as follows:

For certain applications, including many pumps and / or compressors, rotary motion is not required. It is both simple and obvious to connect the end of the reciprocating piston/rod assembly to a pumping or compressing device. However, in many applications it will be preferred for engine final drive to have exclusively rotary motion, requiring a special link between the final drive and any reciprocating plus rotary movement of the piston/rod assembly (effectively the "crankshaft", actually the drive shaft). This can be accomplished by a coupling incorporating either a sliding bearing, such as in a splined propeller shaft, or an assembly incorporating roller, ball, needle or taper bearings. By way of example, Figure 281 shows in cross-section and Figure 282 in elevation a schematic of vehicle-type co-axial nested male 3304 and female 3305 drive shafts capable of reciprocating relative to each other, wherein rotational motion is transmitted via splines 3301 slidably mounted in corresponding grooves 3302. Range of reciprocal motion is indicated at 3303. As another example, Figure 103 shows in cross-section and Figure 104 in elevation a schematic of a coupling between a piston/rod assembly 2078 and a final drive shaft 2079, for applications where loads are transferred in one rotational direction only 2080. Roller bearing races 2081 link planes 2082 inside the piston rod and on the shaft 2083. The connection between the two systems could be anywhere, including inside the piston segment of a piston/rod assembly.

Amend the paragraph beginning "Considering one of the inventions . . ." on page 34 as follows:

Considering one of the inventions in one of its most simplified forms as in Figure 109, we have an upper 3035 and a lower 3036 toroidal combustion chamber in an integral housing system

3007, in which a reciprocating element 3004 also rotates. The extreme surface 3037 of each chamber has a similar folded sinusoidal configuration, as sketched in Figure 110, so arranged that the variation of vertical distance between the two surfaces is the maximum possible. The reciprocating element has a projecting flange 3038 reciprocating in a depression 3038a in the cylinder. The flange is the reciprocating element's (~~(-which is the)~~) working part (it effects compression and transmits expansion forces). The upper and lower surfaces 3039 of the flange are also shaped as in Figure 110, but arranged so that the thickness of the flange is always constant. Because the reciprocating motion is of constant dimension, so the height (distance from peak to valley) of the sine (or similarly shaped) wave will be constant, but the pitch (distance from peak to peak) of the wave will vary, from a maximum at the outer radius of the toroidal combustion chamber, to a minimum at the inner radius. Taking a partial curved cross-section through the two combustion chambers at "A", the path of the reciprocating and rotating flange is sketched in Figure 111, with sine wave height to pitch ratio 1:3 and wherein it is assumed that all four sinusoidal surfaces are identical. The path of a fixed point in / on the flange is indicated at several successive positions marked, a, b, etc. The positions of the flange surfaces at corresponding times are indicated 3039a, 3039b, etc. (The intervals correspond to constant units of rotation.) As can be seen, if all four surfaces are identical the engine would not work (eg the clearance problem in area B). Usually, in any one combustion chamber, the upper surface of that chamber will have to have a different surface from the lower surface of that chamber. Almost any different combination is possible, but often it will involve an upper limit on the theoretically possible compression ratio, since the upper and lower surfaces do not match. At the height / pitch ratio of the sine wave of Figure 112 (1:3), a compression ratio of around 7.5:1 is practicable. If the outermost surface of each chamber retained its sinusoidal form, then a workable form of Section "A" would be as shown schematically in Figure 112. In this case, essentially the flange's valleys would stay more or less sinusoidal, but the peaks would have a sharp apex. If the design compression ratio were less than the theoretical maximum, then it would be possible under constant speed operation for the flange apexes to make no contact with surfaces 3037.

In the paragraph beginning "If the two combustion chambers . . ." on page 36, on the fourth line change "Figure 393" to 'Figure 112'.

Amend the paragraph beginning "Other layouts of guide systems . . ." on page 41 as follows:

Other layouts of guide systems relative to combustion chambers are possible. Figure 125 shows schematically a more powerful engine having four identical combustion chambers 3115 and two identical complete guide assemblies 3116, each having upper and lower tracks. The guides have the same number of reciprocations per revolution, and the engine has to be very carefully assembled, so that the guides are perfectly synchronous with each other and / or the roller assemblies have to be of the type having elastomeric interlayers. Figure 126 and detail Figure 127 show schematically a twin combustion chamber 3115 engine, wherein component 3004 turns clockwise relative to housing 3007, which is mounted on bearings 3120a and itself turns counterclockwise relative to enclosure 3120. Three separate complete twin-track toroidal guide

systems are located at 3117, 3118, 3119. The sine or other waves in each guide system have the same amplitude but differing pitch, so that each system has a different ratio of reciprocation to rotation. Only one system is engagable at any one time, by means of extensible / retractable roller assemblies. Selection of which guide system is engaged is made by movement of ring 3121 (turning at same speed as housing 3007) by means of actuator(s) 3021a. The ring is connected to a series of slidable shafts or elements 3122, which actuate the extension or retraction of the roller assemblies. Preferably the roller assemblies are spring-loaded to the retracted position. Such retractment / engagement devices are known, but the principle is illustrated schematically for a two-speed system in Figure 127, where shaft 3122 has a plate-like section for engagement with a portion of a retractable roller assembly. Sinusoidal tracks may be engagable with non-rotating or solid elements, retractable or fixed.

Amend the paragraph beginning "If one is going to use . . ." on page 42 as follows:

If one is going to use one combustion chamber module to make engines of varying power and swept volume, then the gas passage(s) within the module (if any) should be so sized as to accommodate the gas flows of the largest engines likely to use that module. Figures 129 to 132 illustrate schematically various possible gas flow layouts, wherein 3126 indicates a multiplicity of equal sized toroidal combustion chambers, 3004 the moving component reciprocating in direction 3008, with 3007 the "fixed" housing (which, in all these embodiments, could also rotate), and 3057 an enclosure or casing. "A" represents charge air volume, "B" high temperature and pressure exhaust, "C" lower temperature / pressure exhaust. Filamentary material is shown at 3128a. Porting is not shown, but can be as described elsewhere in this disclosure. Solid arrows describe gas flows through ports, dotted arrows show gas flow to and / or from transfer ports, or flows via passage or plenums as described elsewhere herein. Thermal insulation is indicated (schematically, like all other components) at 3127. In Figure 129, thermal insulation separates charge flow from hot components, charge flows into the combustion chamber, exhaust flows from it into a central exhaust gas reservoir. Obviously, the flows could be reversed, volumes A and B transposed, insulation moved to the interface of component 3004 and the central (now charge) gas reservoir or plenum. Figure 130 shows a system having transfer ports, indicated schematically at 3128. Here again, the flows could be reversed, volumes transposed, insulation repositioned. Figure 131 shows a layout where exhaust gas flows adjacent to the structural component of 3004 and 3007 are used to reduce heat flows (ie thermal gradients) across these components, with the center of the engine occupied by a mechanical system 3130. If 3130 were a fuel delivery system, this could serve to maintain liquid fuel under pressure at temperatures greater than boiling. A compressor and / or turbine system is indicated schematically at 3129 / 3134. In Figures 129, 131 and 132, casing 3057 comprises part of the structure defining volume A, while in Figure 131 thermal insulation 3127 is part of the structure defining volume C.

In the paragraph beginning "The engines of Figures 138 . . ." on page 46, on the third line change "Figure 142" to 'Figure 143'.

Amend the paragraph beginning "In Figure 154 . . ." on page 57 as follows:

In Figure 154 is shown diagrammatically a housing 51 enclosing a reaction volume 52, both having interposed between them and engine 53 with exhaust opening 54, an inter-member 55 of substantially flat configuration. Figure 155 shows a similar arrangement, but with the inter-member 55 in association on one side with both engine 53 and an exhaust opening liner 56, which in the embodiment illustrated is restrained in position by the inter-member 55. Figure 156 shows a similar arrangement to that of Figure 154, but with the substantially flat inter-member 55 recessed into a corresponding depression 59 in the engine 53, being restrained against the block in the embodiment shown by the enclosed housing 51. In Figure 157 is shown an arrangement similar to that of Figure 154, but where the inter-member 58 is of enclosing configuration, that is when viewed in elevation from the reaction volume side it is seen to have a depression 59 defined by a peripheral lip 60, the outline of which corresponds with that of the lip 61 of the enclosed housing 51. A notional plane drawn across the lips will define two sections of the working volume of the reactor, one within the housing at 62, the other within the depression 59, of the inter-member. Figure 158 shows a broadly similar arrangement, but where the mounting between housing and inter-member is used to support filamentary material 63, which is also shown at 63 in Figure 155. Figure 159 shows an arrangement similar to that of Figure 157 (~~(+67)~~), but where the enclosing inter-member 64 has an integral projection 65 on its engine side, in this embodiment of approximately ring or hollow cone like configuration, to act as exhaust opening lining. Figure 160 illustrates the fixing detail at (A) in Figure 154, where an L clamp 66 and bolt 67 press the housing 51 to inter-plate 55 and thence to engine 53. Compressible heat resistant material 68 is interposed between the joints to allow for proper sealing, possible differential expansion of the various components, and more even load distribution between possibly marginally mismatched surfaces. Figure 161 is a detail at (B) of Figure 156 showing a similar fixing technique, and an alternative embodiment where the inter-plate 55 retains in position an exhaust opening liner 56. Figure 162 shows a fixing detail (~~(similar to that)~~) suitable for use at (C) in Figure 158, but retaining a different type of inter-member 69, one which does not substantially mask the engine, but which is part of an effective division of the enclosing housing, the advantages of which are explained below. Here the two portions are shown separately fixed to the block, although in some embodiments only the outer housing need be fixed, depending on detail design. By example, the housing 51 is retained against the inter-member 69, by means of strapping band 70 pivotally attached to winged extensions 71 of a collar 72 mounted on un-threaded portion 73 of a stepped diameter stud 74, by means of nut 75 and washer 76 shown dotted. The inter-member 69 is fixed to the engine 53 by means of the same stud 74, an L clamp 66 and a washer 77 and nut 78 of larger internal diameter than the set 75, 76. Compressible heat resistant sealing material 68 is disposed within the joints between mating surfaces.

Amend the paragraph beginning "By way of example . . ." on page 59 as follows:

By way of example, there is shown in Figure 167 in cross-sectional view and in Figure 168, which is a (~~(#)~~) front elevational view as seen from E, an exhaust opening liner combined with honeycomb configuration gas flow director 83 (~~(-85 and)~~) held in position against engine 53 by inter-member 55. (~~(#there being)~~) There is a heat resistant compressible material 68 between the

joints. Inside the opening 54, the greater mass of gas will be concentrated toward the outside of the curve at 84, and therefore the honeycomb structure has at the end facing the gases a diagonal face 84a across the opening as shown, so that whatever frontal area the honeycomb vanes 85 have will cause the gases by deflection to pass through the structure more evenly distributed. With progression of gas flow the vanes become more mutually further spaced, so reducing gas velocity, and curve away from each other, so that the mouths 86 of the structure will direct the gases in a multiplicity of directions. The honeycomb structure may be of any suitable cross-sectional configuration, including by way of example, that of Figure 169, where the passages have six faces, or that of Figure 170, where the passages are formed by the intersection of radial and coaxial membranes. In an alternative embodiment, gas flow is directed by flanged members running part of the length of the exhaust opening, as shown by way of example in an embodiment illustrated in sectional plan view Figure 171 and in partial cross-section in Figure 172. The flanged members are alternatively "Y" shaped configuration at 87 and of roughly cruciform configuration at 88, and are spaced and held from each other by spacer rings 89 disposed at intervals along the length of the assembly. The flanged assembly of the illustrated embodiment is retained by fitment into grooves 90 in the opening surround 91, such grooves optionally containing a compressible bed 92 at F in Figure 171 and are held against 53 by inter-member 55 sandwiching the bent extension of flanges as at 93 through compressible material 68.

Amend the paragraph beginning "All the features . . ." on page 60 as follows:

All the features described herein may be combined in any convenient or desired way. By way of example, Figure 177 shows a selected embodiment in cross-section. The reaction volume is enclosed by an inter-member 55 of ceramic material having projections comprising exhaust opening liners 56 and spaced from engine by compressible heat resistant material 68 such as ceramic wool, together with an enclosing housing 51 of integral ceramic construction. The joint between the two principal enclosing members supports a filamentary space frame 96 that is a construction of short straight metal rods connected to each other at different angles, which substantially fills the foremost part of the reaction volume, the rearmost portion of which is occupied by filamentary material 18 of wool-like configuration, of say a ceramic based compound. Within the exhaust port area are two metal cone shaped spirals 97 (~~(- the free ends at their cemented back to back meeting projecting to from))~~ mounted back to back with projecting bayonet fixings shown dotted at 98, which locate in grooves 99 running from initial entry away from the direction of the exhaust valve, so that the pressure of gas flow will cause the spring projections or bayonets to seat at the end of the grooves.

In the long paragraph beginning "Another appropriate form of . . ." on page 62, change "Figure 192" to 'Figure 190' and change "Figure 44" to 'Figure 192' , where occurring.

Amend the paragraph beginning "In operation, after . . ." on page 66 as follows:

In operation, after the main valve ((+67)) 166 has closed and junction valve 167 has

opened, exhaust gas will travel down the passage 168 to fill the reservoir 150. A build up of pressure will be caused because the reservoir can only expand against the force of springs 164. The communication between the reservoir and inlet manifold being unobstructed, the gas will escape into the manifold at a rate in proportion to the size of opening and pressure in the reservoir. When the reservoir reaches a point near the limit of its downward expansion (allowance being made for safety margins) the main valve 166 opens, either partly, to maintain pressure if full operating temperature has not been reached, or fully. In the embodiment the aperture between passage 169 and inlet manifold is made very small so that, even under the maximum designed pressure of the exhaust reservoir system 170, the rate of gas flow into the manifold is very low in proportion to flow produced through the exhaust ports, thereby giving a very reduced rate of exhaust gas recirculation. After the reservoir has been filled and gases diverted down the normal exhaust system, the loading of the springs 164 will ensure the slow collapse of the bellows 158 and the continuing bleeding of gas into the inlet system until the reservoir has been emptied. The provision of a second valve communicating with passage 168 may in some configurations be omitted by the provision of a relatively small opening between reactor and passage at junction 167, the opening being of many times smaller cross-sectional area than the main exhaust pipe 170. The smallness of opening will restrict gas flow from reactor during the initial stages of warm-up and main valve 166 closure, until the higher pressure in the reactor accelerates the rate of gas flow along passage 168 to more rapidly fill up the reservoir. The non-closure of the small opening at 167 will ensure that the exhaust gases will effectively be recirculated to the reactor once normal warm operation commences. Depending on the strength of reservoir springs 164, the gas flow rates back through the opening will be lower than those into the reservoir, since the pumping action of the engine must necessarily have considerable greater force than spring action. If it is considered that the gases diverted to the reservoir system have not sufficiently reacted by the time they re-enter the reactor, then catalytic material may be associated with the reservoir, or its internally faced components and / or those of passages 168, 169, or they may be fabricated of a material having catalytic action, such as copper or nickel. Alternatively or additionally, junction 167 may be placed as closely as possible to the exhaust openings, so that the returning gases travel through a substantial portion of the now warm and fully operative reactor. The reservoir assembly may be made of any suitable materials, which to a degree will need to be heat tolerant. If the chosen materials have low heat tolerance, then optional heat dispersal means may be affixed to passage or pipe 168, as shown diagrammatically at 171. If materials are heat resistant, as for example would be a bellows assembly made in silicone rubber, then insulating means may be incorporated on the passages, as shown diagrammatically at 172, with the advantage that the gases may be maintained in the reservoir at warmer temperatures, thereby speeding up reaction processes. The warmth of the gases may be used to advantage in another configuration, where the gases are recirculated to the intake system. The provisions of this flow of warm gas during cold start - as has been shown above, the reactor may be operative to a degree already from a few cycles after firing commences - will assist in vaporization of fuel during engine warm up. In normal usage, the gases will not at inlet entry point be hot enough to present risk of premature fuel combustion. Optionally, a valve 155 (~~((not shown))~~) may be provided between reservoir and inlet system to regulate circulation.

Amend the paragraph beginning "It is desirable to make . . ." on page 69 as follows:

It is desirable to make the valve actuating means as simple and as fail-safe as possible. To this end, the valve should be spring loaded (not locked by mechanical action) in the closed position in such a way that reactor pressure over the designed limit will overcome the force of the spring sufficiently to allow some gas to escape, thereby again lowering pressure to below that required to actuate the spring and maintaining a balance of loading to keep the valve slightly open, to sustain constant pressure in the reactor. The spring loading is such to also bias the valve to the fully open position. Such an arrangement is illustrated by example diagrammatically in Figure 242, where 210 shows a valve actuating lever in heavy line, butterfly valve 211 and internal face of passage 212 in light line, spring 213, spring axis 214 and spring anchorage 215 on housing and anchorage 216 on lever, with pivotal valve axis at 217. The valve assembly is shown in slightly open position in dotted line and fully open in chain dotted line, with dashed line 211a indicating the arc of valve travel. The same system of loadings may be employed and the valve actuated by making the previously fixed spring anchorage point 215 movable as in the path indicated by dashed line 218 between extremities 219 and 220, dashed line 214 indicating spring axes at each extremity. This movement of spring anchorage may be actuated in any way, and in a selected embodiment is moved by a member driven by the expansion of heat sensitive material, such as a trapped pocket of gas or as is shown in Figure 243, where a piston 221 communicates with a container of high conductivity 222 exposed to the passage of hot exhaust gas 223 through a volume 224 of trapped readily expansible material such as gas or wax. The piston 221 is connected to rod 225 and linkage 226. Figure 244 shows schematically how the piston rod 225 actuates the operation of the valve by means of its actuating lever 210, spring 213, and an intermediate arm-shaped lever 227, mounted on pivot 228. The actuation of the valve indirectly, by means of a spring, ensures that fail-safe characteristics are embodied. If this is not considered necessary, then the heat actuated piston 221 may by direct linkage open and close the valve, as for instance if the end 229 of the intermediate lever 227 were connected directly to the valve actuating arm (embodiment not illustrated). In both cases, but especially in the latter, it will be possible to closely relate valve opening to exhaust temperature, and therefore reactor pressure in relation to temperature.

Amend the paragraph beginning "The above features . . ." on page 70 as follows:

The above features may be used in any suitable combination with each other and also, where appropriate to fulfill functions not related to cold start. Gas circulation to inlet system may be associated with a gas reservoir, or alternatively it may be direct, that is eliminating the reservoir. Further, the exhaust gas recirculation (ERG) system described previously could for example be used after warm up had been achieved to provide EGR to the engine under normal running, either continuously or under certain operating modes. To facilitate the use of EGR, and so thereby possibly to eliminate the use of pumps, a scoop may be placed in the reactor about the junction with recirculation passage, as illustrated diagrammatically in Figure 245, where the scoop 230 projects into the exhaust gas flow 231, so creating a higher pressure area at 232, which assists the flow of gas along the EGR system 233. Preferably, the scoop is placed in a "weak" area of the reactor, that is where the reactions are taking place at below average rates, so that the least pollutant free gases are recirculated, permitting the reactions partly to continue during their second passage through the reactor. The scoop arrangement would entail that EGR employed

continuously is in roughly constant proportion, after a build up of proportion between very low and medium speeds, since gas circulated depends on speed and therefore volume of gas issuing from the engine. ~~((Generally EGR absorbs engine power but, at certain lower rates and / or operating conditions, EGR may marginally increase engine power. For this reason, and / or to better eliminate pollutants, it may be desired to have EGR operative under only specific running conditions, such as acceleration or deceleration, etc.))~~

In the paragraph beginning "An optional valve at . . ." on page 70, in line 2 change "Figure 90" to 'Figure 246'.

Amend the paragraph previously beginning "It has been said . . ." on page 71 as follows:

~~((It has been said that EGR may under certain conditions contribute to marginal increases in power. In fact it is almost impossible for this to be achieved directly; any power gains are caused by the reduction of octane number requirement that EGR results in, thereby permitting increased compression ratios and more optimum valve and ignition timing for a given fuel. Because EGR assists in the prevention of pre-detonation or "knock," it is usually required especially at high load conditions. Previous systems have been proportioned to inlet vacuum, which is not necessarily very great under all high load situations. At least a portion of the EGR system, preferably under low pressure perhaps maintained by a reservoir, may therefore be connected directly to an enriching circuit in a fuel supply only operative under high load conditions. Alternatively))~~ In situations where EGR may desirable at moderate to higher engine speed, an inlet gas velocity actuated valve, as shown in section plan Figure 247 and elevation Figure 248, may be incorporated at the junction of EGR system to inlet manifold. The valve, shown open in Figure 247, comprises a shaft 243 slidable in a passage 244 communicating with EGR system, exposing a progressively sized vent 245, said shaft terminating in a head 246 having attached to it scoops or vanes 247 projecting into the gas stream 248 against the action of looped leaf spring 249. Figure 248 shows the same arrangements with the valve, which is accommodated in a housing 250 projecting clear of inlet manifold wall 251, in the closed position. Preferably a properly balanced EGR system will comprise a series of valves, say actuated by vacuum and / or velocity or other means, disposed in different parts of the inlet system and all communicating with the EGR system, preferably having a gas reservoir. By careful positioning of these valves, regulation of their spring bias ~~((to closed))~~ and selection of passage diameter, the right amount of EGR could be provided for the various driving modes.

Amend the paragraph beginning "Described below are . . ." on page 72 as follows:

Described below are means of introducing substances to an intake charge which do not involve the vaporization of fuel by gas velocity. Any of these means may be employed for the introduction of both the secondary substance and / or the main fuel to the charge. In the case of compression ignition engines or other engines having cylinder primary fuel injection, the other substances may be supplied by means of additional injectors, or they may be introduced by

compound injectors, that is by different passage systems in the same injector. The injection may be linked, that is injection of one substance will automatically cause the introduction of another, or the systems may operate independently of one another. Figure 252 shows by way of example a diagrammatic section of the lower portion of an injector where the primary fuel 272 is injected in the normal way at 273 by the lifting of nozzle 274, which has a hollow central passage 275 linking with a secondary fuel gallery at 276 only when nozzle lift and consequently normal fuel injection is taking place. The secondary fuel is under continuous pressure and will therefore inject at 277 only when nozzle lift occurs. The proportion of normal to secondary fuel is determined by their respective pressures and the duration of degree of overlap between gallery and hollow passage. Figure 253 shows diagrammatically a compound injector having an inner nozzle 278 coaxial and within an outer nozzle 279, both operating in the conventional mode with independent lift and injection capacity. This has the possible disadvantage of the long fuel travel down the hollow passage of the central nozzle. By way of example, a design involving a shorter central nozzle fuel travel from pressure reservoir to tip is shown schematically in cross-section in Figure 254 and in plan in Figure 255, where the nozzle assembly is viewed from the combustion volume. The central nozzle 280 operates in the conventional manner, moving vertically on its axis in the release of fuel, while the outer nozzle 281 moves coaxially on the first and in its seating in a rotational mode during the release of fuel. The rotational movement is imparted against the resistance of friction seals 282 by means of jets 283 terminating tangentially to diameter of nozzle, so imparting to it a twisting motion due to the force of, and for the duration of, fuel injection. This will result in a slinging of fuel across the combustion volume in the manner indicated at 284, in a similar manner to the action of some garden hoses. The injection of the outer nozzle is effected by means of a pressure wave in the coaxial and surrounding fuel chamber 285, which will depress one or more plungers 286 against spring 287 loading, and so by inward movement mate up fuel galleries to make connection and allow for fuel travel between the chamber 285 and jet 283 tip. The jet 283 has been called such to distinguish it from nozzles proper as at 280 and 281. This slinging action imparted by rotational nozzle movement, the latter in turn imparted by the tangential direction of fuel spray, has considerable benefits over conventional injection systems. The latter operate in straight line distribution of fuel, while the snakelike shape formed by the spray of invention is of greater length, thereby lessening the chance of liquid deposition or combustion in chamber walls before atomization has taken place. The slinging action also tends to distribute the droplets of fuel through a greater volume of charge than the conventional unidirectional injection.

In the paragraph beginning "Generally in the previous embodiments . . ." on page 78, change "Figure 286" to 'Figure 277'.

Amend the paragraph beginning "A feature of the invention is . . ." on page 80 as follows:

A feature of the invention is the provision of a variable diameter charge intake throat. This may be used with any type of engine, but preferably forms charge entry point to the housing of the invention. Essentially the variable throat comprises a stretched elastomeric tube about which is wound one or more tension members, whose free ends once pulled effect a reduction in tube diameter. Section plan view Figure 278, cross-sectional view Figure 279 and detail Figure 280 show

diagrammatically a stretched rubber throat shown solid in open position at 739, fixed within charge housing 740 by means of expandable clamp rings 741. Wound externally about the elastic throat 739 and mounted in lubricant 743 in guide channels 742 are multiple tension members 744 of nylon (shown in detail section Figure 280), whose ends are taken via pulleys 745 and wound about variable diameter cylinder 746 mounted adjacent to throat. In operation, rotation of cylinder causes the tension members to effect a partial strangulation of throat, so reducing its diameter, as shown dashed in Figures 278 and 279. It is desirable that throat or membrane 739 when in the open position should be in significantly greater tension due to stretching in direction 747 than in direction 748, this differential ensuring throat remains open.


IN THE DRAWINGS:

Generally, where the drawings have been too small, too difficult to read easily or too crowded, they have been enlarged, clarified or spaced further apart. Incorrect Figure numbering has been corrected. Feature identify numbers have been corrected, or added where necessary, and changed to the correct size. In a few cases, where previously a feature was just indicated a rectangular line (identified or described in the text), a little more illustration has been added to make it easier to visually identify that feature, without referring to the text.

End of Amendment.

Please debit my patent office deposit account number **501 334** if any fees should be due.

Sincerely,

A handwritten signature in black ink, consisting of a series of loops and a long horizontal stroke at the end.

Mitja Victor Hinderks.

Sole inventor, applicant and power-of-attorney of record.